ASME Paper Tile: GT-ACYSS: gas turbine arekret-cycle simulation modelling for training and educational purposes

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ASME Journal Title: Journal of Nuclear Engineering and Radiation Science,

Volume/Issue: Vol. 5, Iss. 4, Article id 041207 Date of Publication (VOR* Online) 2 August 2019

ASME Digital Collection URL: http://nuclearengineering.asmedigitalcollection.asme.org/article.aspx?articleid=27332585

DOI: https://doi.org/10.1115/1.4043681

VOR (version of record)

Journal of Nuclear Engineering and Radiation Science.
Received January 5, 2019;
Accepted manuscript posted May 8, 2019.
DOI:10.1115/1.4043681


Volume 5, Issue 4, October 2019
Article number 041207

# GT-ACYSS: Gas Turbine Arekret-Cycle Simulation Modelling for Training and Educational Purposes

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**This paper presents the modelling approach of a multi-purpose simulation tool called GT-ACYSS; which can be utilized for the simulation of steady-state and pseudo transient performance of closed-cycle gas turbine plants. The tool analyses the design point performance as a function of component design and performance map characteristics predicted based on multi-fluid map scaling technique. The off-design point is analyzed as a function of design point performance, plant control settings and a wide array of other off-design conditions. GT-ACYSS can be a useful educational tool since it allows the student to monitor gas path properties throughout the cycle without laborious calculations. It allows the user to have flexibility in the selection of four different working fluids, and the ability to simulate various single-shaft closed-cycle configurations, as well as the ability to carry out preliminary component sizing of the plant. The modelling approach described in this paper has been verified with case studies and the trends shown appeared to be reasonable when compared with reference data in the open literature, hence, can be utilized to perform independent analyses of any referenced single-shaft closed-cycle gas turbine plants. The results of case studies presented herein demonstrated that the multi-fluid scaling method of components and the algorithm of the steady state analysis were in good agreement for predicting cycle performance parameters (such as efficiency, and output power) with mean deviations from referenced plant data ranging between 0.1% and 1% over wide array of operations.**

Keywords: Simulation, modelling, closed-cycle gas turbine, corrected parameters, steady state, single-shaft

## Nomenclature

**Notations**

| | |
|---|---|
| *C* | *heat capacity, (J/K)* |
| **$C^*$** | heat capacity rate ratio, (nondimensional) |
| $C_p$ | *specific heat of gas at constant pressure, (J/kg K)* |
| *CW* | *compressor work, (W)* |
| *CMF* | *corrected mass flow, (kg/s)* |
| **CMSF** | corrected mass flow scaling factor |
| **CSSF** | corrected speed scaling factor |
| **CH** | corrected enthalpy drop |
| **CS** | corrected speed |
| **DH** | work function |
| **DHSF** | work function scaling factor |
| *EGT* | *exhaust gas temperature, (°C)* |
| *H* | *specific enthalpy, (J/kg)* |
| *N* | *rotational speed, (rpm)* |
| **NDCW** | non-dimensional compressor work |

| | |
|---|---|
| **NDTW** | non-dimensional turbine work |
| **NSF** | corrected speed scaling factor |
| **NTU** | number of transfer unit |
| *P* | *pressure, (Pa)* |
| $P_{ref}$ | *reference pressure, (Pa)* |
| **PR** | pressure ratio |
| **PRSF** | pressure ratio scaling factor |
| $Q_g$ | *heat gained, (W)* |
| $Q_{actual}$ | *actual heat flux,* $(W/m^2)$ |
| $Q_{max}$ | *maximum heat flux,* $(W/m^2)$ |
| *R* | *specific gas constant, (J/kg K)* |
| *S* | *specific entropy, (J/kg K)* |
| *SOP* | *shaft output power, (W)* |
| *SP* | *specific power,* $(W/kg\ s^{-1})$ |
| *T* | *temperature, (℃)* |
| $T_{ref}$ | *reference temperature, (℃)* |
| *TET* | *turbine entry temperature, (℃)* |
| *TW* | *turbine work, (W)* |
| *UW* | *useful work, (W)* |
| *W* | *mass flow, (kg/s)* |

**Greek Symbols**

| | |
|---|---|
| $\gamma$ | ratio of specific heat of gas |
| $\varepsilon$ | effectiveness |
| $\eta$ | efficiency |
| $\eta_m$ | mechanical efficiency |
| $\eta_{th}$ | cycle thermal efficiency |
| $\Delta$ | difference |
| $\delta$ | ratio of design pressure to reference pressure |
| $\theta$ | ratio of design temperature to reference temperature |

**Subscripts**

| | |
|---|---|
| *c* | *compressor* |
| $c_{in}$ | *compressor inlet* |
| *cold* | *fluid cold stream* |
| $c_{out}$ | *compressor outlet* |
| *Dp* | *design point* |
| *Dprefmap* | *design point reference map* |
| *HEX* | *heat exchanger* |
| *hot* | *fluid hot stream* |
| *HPS* | *high-pressure side* |
| *in* | *inlet condition* |
| *LPS* | *low-pressure side* |
| *m* | *mechanical* |
| *max* | *maximum* |
| *min* | *minimum* |
| *OD* | *off design point* |
| *out* | *outlet condition* |
| *refmap* | *reference map* |
| *t* | *turbine* |
| $t_{in}$ | *turbine inlet* |
| $t_{out}$ | *turbine outlet* |
| *1-7* | *station number* |

**Abbreviations**

| | |
|---|---|
| *A* | *working fluid coefficient* |

| | |
|---|---|
| *ACYSS* | *Arekret cycles simulation* |
| *CBC* | *closed Brayton cycle* |
| *DP* | *design point* |
| *FORTRAN* | *IBM patent computer programming language name* |
| *GH* | *gas heater* |
| *GT* | *gas turbine* |
| *HP* | *high pressure* |
| *HPC* | *high pressure compreesor* |
| *HTGR* | *high-temperature gas reactor* |
| *i* | *index* |
| *IC* | *inter-cooler* |
| *JAEA* | *Japan Atomic Energy Agency* |
| *LP* | *low pressure* |
| *LPC* | *low pressure compressor* |
| *MW* | *molecular weight of selected gases* |
| *NASA* | *national aeronautics and space administration* |
| *OD* | *off-design* |
| *PC* | *pre-cooler* |
| *ReX* | *recuperator* |
| *SF* | *scaling factor* |
| *TURBOMATCH* | *tradename of computer code developed at Cranfield Univesity* |

## 1. Introduction

With improved modelling techniques, simulations have assumed an essential role in planning, executing and evaluating operations, in order to reduce acquisition costs, increase system performance and improve maintenance. Gas turbine (GT) engine simulation can be valuable tools for teaching in an academic environment or for on-the-job training for engineers within the industry. Simulation tools can assist in the task of explaining, by making it possible to observe what is happening within each individual component and overall engine behaviour [1].

Consequently, a great deal of research work has been expended in developing analytical tools capable of evaluating the steady-state and dynamic performance of gas turbine engines such as references [2–6]. However, most of the documented models in open literature [4,5,7,8] were developed mainly for open-cycle gas turbine operations or for specific design point closed-cycle power plant projects which may not be accessible for commercial use or allow for complex off-design engine configuration simulation. However, the focus of this paper is on closed-cycle gas turbine simulation. This is so because of the renewed interest in closed-cycle gas turbine technology development for nuclear, marine, or space power applications, due to its performance advantages at part-load, environmental compatibility, reliability and flexibility in working fluid usability.

Broadly speaking, a review of the available simulations model for closed-cycle gas turbine; both analogues and non-linear digital have been discussed by references [7,9]. Vavra [10] described a method that graphically illustrated the fundamentals of matching components in the closed-cycle gas turbine. The reports of Dostal [11] and Dyreby [12] documented various aspects of simulations program for Supercritical Carbon Dioxide (S-CO2) closed-cycle gas turbine plants. Essentially, their approach consists of synthesizing the thermodynamic relationships that described each engine component. Dynamic equations describing the transient behaviour of the High-Temperature Gas Reactor coupled to a gas turbine (GT-HTGR) have been documented in references [13–15], and Dupont et al. [16], described a computer code called TugSim-10 developed for transient simulation of nuclear power plants. The review by Olumayegun [9] covered various aspects of simulations for closed-cycle gas turbines.

From the review of published works cited so far, it would appear that the modelling tools mentioned above were developed for either open-cycle gas turbine engine configurations or for only specified design point closed-cycle gas turbine power plant projects, or specific configuration and specific working fluid applications. There is still need for a general model that may be used to represent different closed-cycle gas turbine engine configurations, flexibility in working fluid selection capability and the ability to study the engine behavior at the conceptual design stage; considering the effect of component performance on the overall performance of the engine, and capability for complex off-design simulations, and preliminary component designs.

This paper introduces Gas Turbine Arekret Cycle Simulation (GT-ACYSS), a program developed by the authors for closed-cycle gas turbine steady-state performance simulation and preliminary component designs. It describes the structure of the code, computational models used, and verifications based on the open literature of reference power plants used in case studies. The code allows the choice of key design point parameters and computes the cycle performance at different off-design conditions, e.g. at varying temperature ratios, pressures, rotational speeds and part load. The code also allows the user to simulate different working fluid namely; helium, water, air, nitrogen, and carbon dioxide, to give the user a valuable understanding on how these working fluids could affect performance and component design decisions. This code can be a useful educational tool since it allows the student to monitor gas path properties throughout the closed-cycle configuration without laborious calculations. Another important benefit of the code is the flexibility it gives the user to understand the different closed-cycle gas turbine control strategies by selecting any specific control mechanism mentioned in section VII during off-design or pseudo transient simulation.

Therefore, the GT-ACYSS model presented is envisioned as building blocks for more complex shaft configurations, more working fluid selection options and for flexibility in meeting specific simulation needs.

## 2. Description of GT-ACYSS Simulation Tool

The code was developed in FORTRAN 90 and enables the performance and component sizing analysis of the single-shaft closed-cycle gas turbine. The main objective of its development was to produce a user-friendly program for closed-cycle gas turbine performance simulation to support teaching and training purpose like the TURBOMATCH code [1,2], already developed in Cranfield University to enhance performance simulation capabilities for open cycle gas turbine configurations. The code was developed based on conceptual division of the engine into its components as shown in fig. 1.

The overall program structure shown in fig. 1 consists of modules and subroutines that inter-connect to simulate the steady-state performance of the listed engine configurations. The major advantage of this kind of structure is that it creates flexibility in case of any further modifications. The function files listed have been briefly described in Table 1.

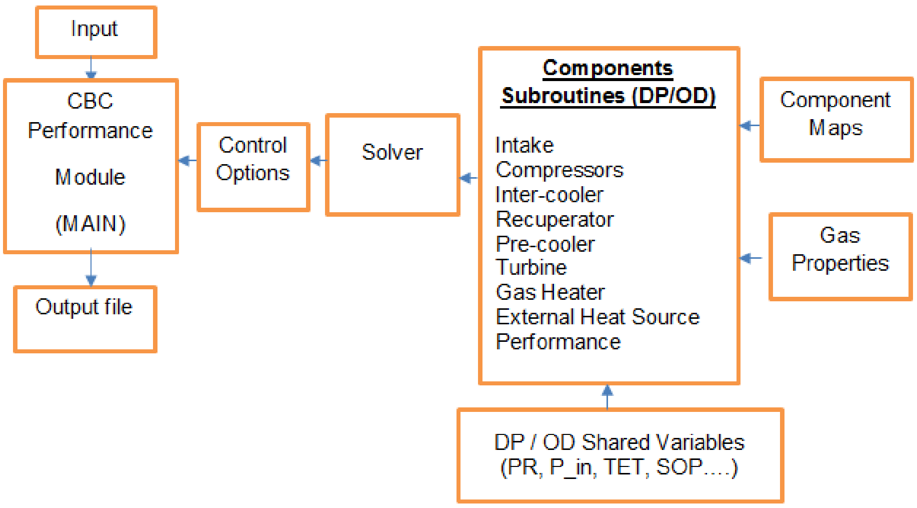


**Fig. 1 Structure of GT- ACYSS: Closed-Brayton Cycle Gas Turbine Simulation Code**

**Table 1. Summary of GT- ACYSS Modelling File Functions**

| List Name | Class Name | Filename | Description |
|---|---|---|---|
| DP/OD shared Variables | Module | shared_variables.f90 offdesign_variables.f90 | Contains list of defined shared variables for Design, Off-design Point and Components sizing (P,W, T etc) |
| Control Options | Subroutine | Control_files.f90 | Defines the engine configuration, control handle, working fluid, & calculations |
| CBC Main | Program | CBC_main.f90 | Part of the code which calls all modules and subroutines |
| Component | Module | Component_name.f90 | Contains components performance and sizing calculations |
| Input and Output | Import/export files | name.dat, name.txt | Contains input files and writes output files |
| Solver | Subroutine | Design_Point_Solver Offdesign_Solver.f90 | Contains the off design point solver based on analytical or component map calculation. Calls all component modules based on cycle configuration |
| Gas properties | Subroutine | Gas_properties.f90 | Contains all the fluid properties model |
| Component Maps | Subroutine | Compressor_map.f90 Turbine_map.f90 | Contains component maps and does scaling calculations |

## 3. Working Fluid Options

GT-ACYSS allows the user to make a selection from different working fluid namely; helium, air, nitrogen, and carbon dioxide. The fluid properties were provided by a series of gas properties subroutine which computes enthalpy, specific heat capacity, viscosity, specific heat capacity ratios and other properties of the fluid.

### A. Modelling of Fluid Properties in GT-ACYSS

The correct simulation of the thermodynamic properties of the working fluid is a major premise for the accurate modelling of the performance simulation code. An estimation of the properties of working fluid listed in GT-ACYSS was modelled using empirical correlations and coefficients. The empirical equations and coefficients are power expansions in terms of pressure and temperature. They were obtained experimentally by fitting the power series to measured isotherm and isobar from experimental data and tables of references [17–20]. A modification on the curve fitting equation as in reference [21] was used to ensure that the coefficients approximations implemented in this tool were able to predict the behaviour of the working fluid at different pressure and temperature in comparison with the tables of reference [20]. For $CO_2$, its properties change rapidly with pressure and temperature near its critical points, especially at the low-temperature region. Similarly, all working fluids were modelled as a function of pressure and temperature. The results of the curve-fitted data were compared against the NASA SP-273 [19], and gas tables of reference [20]. A general representation of polynomial for some of the gas properties is shown in equations (1) – (6).

1) Specific Heat at Constant Pressure

$$C_p = \sum_{i=0}^{i=6} A_i T^i \; for \; T < 127°\text{C} \tag{1}$$

$$C_p = \sum_{i=0}^{i=6} i A_i T^{(i-1)} \quad for \;\; T \geq 127°\text{C} \tag{2}$$

Whereas,

$$A_i = f\,(P)$$

2) Enthalpy: The enthalpy of the working fluids were calculated using integration and averaging as shown in equation 3

$$H = \int C_p dT \tag{3}$$

3) Entropy

$$S = S_0 + A_0 lnT + \sum_{i=1}^{i=6} A_i \frac{T^i}{i} \; for \; T < 127℃ \tag{4}$$

$$S = S_0 + A_0 lnT + \sum_{i=2}^{i=6} \frac{i}{i-1} A_i T^{(i-1)} \; for \; \geq 127℃ \tag{5}$$

Whereas,

$$S_o, A_o, A_i = f\,(P)$$

4) Gamma

$$\gamma = \frac{C_p}{(C_p - R_{gas})} \tag{6}$$

*Where* $R_{gas} = \frac{R}{MW}$, *R = 8314.36, MW = Molecular Weight of Individual selected gases.*

An example of the correlation coefficient utilized for $CO_2$ can be found in the appendix.

## 4. Engine Configuration Options

A computer simulation program should allow the user to assemble the component in the desired configuration [4,5,22]. Each component has its own distinct performance characteristics, which may be modelled either analytically or by experimentally obtained performance data. The program should be modular so that if desired, an individual module may be used to study the performance of the components comprising of a particular engine configuration.

Similar to the open cycle gas turbine, a closed-cycle gas turbine may either have a single or multi shafts arrangement of simple, recuperated, or intercooled cycle configuration. At the moment, this code allows for only single-shaft, simple, recuperated, intercooled, and intercooled-recuperated configuration. Since the program is modular in structure, other cycle configuration will be incorporated to it in the future.

## 5. Components Simulation Modelling

The overall performance of the closed-cycle gas turbine is determined by gas path properties and balance of its components. The mathematical model which describes the physical behaviour of the system components were created by governing analytical and empirical data described in [23–27].

### A. Compressor

The behaviour of the compressor is described by dimensionless parameters. The performance characteristics are usually plotted as pressure ratio against dimensionless or Corrected Mass Flow (CMF), speed (CS) and enthalpy (CH). For a particular working fluid, the dimensionless parameters can be represented by the following expressions.

$$CMF = \left(\frac{W\sqrt{\theta}}{\delta} \times \sqrt{\frac{R}{\gamma}}\right), \qquad CS = \left(\frac{N}{\sqrt{\theta R \gamma}}\right), \qquad CH = \left(\frac{\Delta H}{\sqrt{\theta R \gamma}}\right) \tag{7}$$

Where,

$$\theta = \frac{T}{T_{ref}}, and\ \delta = \frac{P}{P_{ref}}$$

The temperature at the exit of the compressor is obtained from the inlet temperature, isentropic efficiency, pressure ratio and ratio of specific heats, given by the expression

$$Tc_{out} = T_{c_{in}} + \frac{T_{c_{in}}}{\eta_c}\left[\left(\frac{P_{c_{out}}}{P_{c_{in}}}\right)^{\left(\frac{\gamma-1}{\gamma}\right)} - 1\right] \tag{8}$$

Where the compressor discharge pressure is derived from the given pressure ratio

$$PR_c = \frac{P_{c_{out}}}{P_{c_{in}}} = f\ (CMF, CS) \tag{9}$$

The corrected compressor work (NDCW) is derived as

$$NDCW = \left(\frac{CW}{\delta\sqrt{\theta}}\right) = f\ (CMF, CS) \tag{10}$$

Where CW is the compressor work, which is a product of the mass flow rate, specific heat capacity at constant pressure and the overall temperature rise in the compressor.

$$CW = WC_{P_{out}}T_{c_{out}} - WC_{P_{in}}T_{c_{in}} \tag{11}$$

In order to solve Eqs. (7)- (11), the needed input data can be obtained from the design point inlet conditions and compressor performance map.

**B. Turbine**

Similar to the compressor, the performance characteristics of the turbine is described by a number of dimensionless parameters. The non-dimensional parameters described for the compressor is similar to the turbine. The input data at the design point are the turbine inlet conditions (mass flow rate, temperature, and pressure), and component efficiency

The temperature at the outlet of the turbine is obtained from the following expression:

.

$$T_{t_{out}} = T_{t_{in}} - T_{t_{in}}\eta_t\left[1 - \left(\frac{P_{t_{out}}}{P_{t_{in}}}\right)^{\left(\frac{\gamma-1}{\gamma}\right)}\right] \tag{12}$$

The corrected turbine (NDTW) work can be obtained from

$$NDTW = \left(\frac{TW}{\delta\sqrt{\theta}}\right) = f\ (\frac{P_{t_{out}}}{P_{t_{in}}}, CS) \tag{13}$$

Where TW is the turbine work, which is expressed as:

$$TW = WC_{P_{out}}T_{t_{out}} - WC_{P_{in}}T_{t_{in}} \tag{14}$$

Implementing system components pressure losses will mean that the turbine pressure ratio becomes;

$$PR_t = \frac{P_{t_{out}}}{P_{t_{in}}} = PR_c \left[\frac{\sum(1 - \Delta P)_{HPS}}{\sum(1 + \Delta P)_{LPS}}\right] \tag{15}$$

## C. Heat Exchangers

The heat exchangers which include the recuperator, intercooler, gas heater and pre-cooler were modelled using the ε-NTU method and a counter-flow shell and tube configuration was assumed. The ε-NTU method was used since the inlet condition (temperature and pressure) of the fluid stream can be easily obtained and simplifies the iteration involved in predicting the performance of the flow arrangement. This method is fully described in references [28,29] and has been applied to complex counter flow heat exchanger by Navarro [30]. The approach also assumes that the heat exchanger effectiveness is known and the pressure losses are given.

Therefore, the effectiveness of the heat exchanger is the ratio of the actual heat transfer rate to the thermodynamically limited maximum heat transfer rate available in a counter flow arrangement.

$$\varepsilon = \frac{Q_{actual}}{Q_{max}} = \frac{C_{hot}(T_{hotin} - T_{hotout})}{C_{min}\,(T_{hotin} - T_{coldin})} = \frac{C_{cold}(T_{coldout} - T_{coldin})}{C_{min}\,(T_{hotin} - T_{coldin})} \tag{16}$$

Where $C_{min}$ and $C_{max}$ are the smaller and larger of the two magnitudes of $C_{hot}$ and $C_{cold}$

$$C_{hot} = (WC_P)hot\ fluid\ Stream, \qquad C_{cold} = (WC_P)cold\ fluid\ Stream$$

$$C_{min} = \begin{cases} C_{hot}\ for\ C_{hot} < C_{cold} \\ C_{cold}\ for\ C_{cold} < C_{hot} \end{cases} \tag{17}$$

For counter flow shell and tube heat exchangers, Number of Transfer Unit (NTU) is given by:

$$NTU = \frac{LOG_e \left[\frac{2 - \varepsilon(1 + C^* - \eta_{Hex}}{2 - \varepsilon(1 + C^* + \eta_{Hex}}\right]}{\eta_{Hex}} \tag{18}$$

Where

$$C^* = Capacity\ rate\ ratio = \frac{C_{min}}{C_{max}} \tag{19}$$

$$\eta_{Hex} = (C^{*2} + 1)^{0.5} \tag{20}$$

Assuming Pressure losses, then the exit pressure at design point is given as;

$$P_{out} = P_{in}(1 - \Delta P) \tag{21}$$

Where $\Delta P$ is the percentage of pressure loss specified at the input by the user on each component of the heat exchangers. It is important to emphasis, that GT-ACYSS allows the user to specify whether the pre-cooler utilizes water cooling or dry cooling in order to be able to assess the cogeneration capability of the plant modelling in future applications. The code was modelled to have the working fluid temperature after pre-cooling process equal to the compressor entry temperature; although in a real scenario the temperature may vary a little.

## D. Heat Source Model

In GT-ACYSS, the energy input to the system is modelled as an external heat source supplying thermal heat input at a specified temperature and combustion efficiency. The model allows the user to assume the type of heat source (nuclear, fossil or solar).

Therefore, heat gained is given by:

$$Q_g = W(C_{P_{out}}T_{out} - C_{P_{in}}T_{in}) \tag{22}$$

Calculating for a given total pressure loss is similar to Eqs. (21).

**E. Engine Performance Calculation**

The overall plant cycle assessment is represented as Shaft Output Power (SOP), Specific Power (SP), and cycle thermal efficiency. These are given by the following equations:

$$SOP = TW - CW/\eta_m \tag{23}$$

The capacity of the plant is represented as specific power (SP), given by:

$$SP = SOP/W \tag{24}$$

The cycle thermal efficiency is given by:

$$\eta_{th} = SOP/Q_g \tag{25}$$

## 6. Components Matching

The interaction of the gas turbine components is often referred to as component matching. To be able to predict an accurate design and off-design point performance of the closed-cycle gas turbine would require an accurate assessment of the component performance, sizing and matching of both the turbomachinery and heat exchangers, which is accomplished through details of their maps or analytical models. These operating maps give detailed illustrations at a wide range of operating conditions for both design and off-design points. The interaction of the gas turbine components defines the operating restrictions to achieve satisfactory performance under different conditions. The procedure used in GT-ACYSS was based on references [10,24,26,27]. The components matching were analyzed in line with mass flow compatibility, speed compatibility and work compatibility. A typical illustration using engine model shown in fig.3 is represented by equations (26)-(30) to describe the matching requirements.

Mass flow compatibility between turbine and compressors is expressed as:

$$\frac{W_8\sqrt{T_8}}{P_8} = \frac{W_4\sqrt{T_4}}{P_4} \times \frac{P_4}{P_5} \times \frac{P_5}{P_8} \times \sqrt{\frac{T_8}{T_4}} \times \frac{W_8}{W_4} \tag{26}$$

The High Pressure (HP) compressor inlet temperature $T_4$ in fig. 3 is controlled by the inter-cooler, and is assumed to remain approximately constant due to intercooling. Therefore, the operating line on the HP compressor characteristics will tend towards surge along the line of constant non-dimensional speed as the load demand increases. Hence, the HP compressor inlet non-dimensional flow will also be approximately constant. The flow compatibility between the Low Pressure (LP) compressor exit and the HP compressor inlet is given by:

$$\frac{W_4\sqrt{T_4}}{P_4} = \frac{W_3\sqrt{T_3}}{P_3} \times \frac{P_3}{P_4} \times \sqrt{\frac{T_4}{T_3}} \times \frac{W_4}{W_3} \tag{27}$$

Similarly, the flow compatibility between the LP compressor inlet and exit can be expressed as:

$$\frac{W_3\sqrt{T_3}}{P_3} = \frac{W_2\sqrt{T_2}}{P_2} \times \frac{P_2}{P_3} \times \sqrt{\frac{T_3}{T_2}} \times \frac{W_3}{W_2} \tag{28}$$

The Compressor shaft speed equal turbine shaft speed as shown:

$$\frac{N_8}{\sqrt{T_8}} = \frac{N_2}{\sqrt{T_2}} \times \sqrt{\frac{T_2}{T_8}} \tag{29}$$

The Work compatibility for which (Turbine work > Compressors work) is given by:

$$UW = TW - (CW1 + CW2) \tag{30}$$

## 7. Multi-Fluid Map Scaling Technique

One major challenge to the accurate simulation of any closed-cycle engine model for a new entrant or existing power plant is the difficulty in accessing component characteristics maps; hence, researchers often rely on scaling methods to develop theoretical component maps from existing maps. The maps for different components utilized in this program were obtained using multi-fluid scaling methods which multiplies the scaling factors derived at design point to the original component map at the off-design point.

The referenced map used in GT-ACYSS was adapted from TURBOMATCH; a Cranfield University in-house performance simulation tool [3]. To accommodate working fluid flexibility, maps were scaled considering the influence of gamma and specific gas constants. The scaling factor utilized in the program were obtained using equations (31) – (35), and are documented in references [21,27,31,32].

The corrected mass flow scaling factor (CMSF) is given by:

$$CMSF = \frac{\left(\frac{W_{Dp}\sqrt{\theta_{Dp}}}{\delta_{Dp}} \times \sqrt{\frac{R_{gas}}{\gamma_{gas}}}\right)_{DP}}{\left(\frac{W_{Dp}\sqrt{\theta_{Dp}}}{\delta_{Dp}} \times \sqrt{\frac{R_{gas}}{\gamma_{gas}}}\right)_{DPrefmap}} \tag{31}$$

The corrected speed scaling factor (NSF) can be obtained from:

$$NSF = \frac{\left(\frac{N}{\sqrt{\theta_{Dp}\gamma_{gas}R_{gas}}}\right)_{DP}}{\left(\frac{N}{\sqrt{\theta\gamma R}}\right)_{DPrefmap}} \tag{32}$$

The pressure ratio scaling factor (PRSF) is expressed as:

$$PRSF = \frac{(PR_{DP} - 1)}{(PR_{DPrefmap} - 1)} \tag{33}$$

Isentropic efficiency scaling factor is given by:

$$\eta_c SF = \frac{(\eta_c)_{DP}}{(\eta_c)_{DPrefmap}} \tag{34}$$

The work function scaling factor is:

$$DHSF = \frac{\left(\frac{\Delta H}{\sqrt{\theta\gamma R}}\right)_{DP}}{\left(\frac{\Delta H}{\sqrt{\theta\gamma R}}\right)_{DPrefmap}} \tag{35}$$

The scaling factors are calculated at the design point, hence, there no changes at off-design conditions.

Hence, the corresponding data points for the new map are obtained using equations (36) – (39)

$$Pressure\ ratio\ (PR) = \frac{(PR_{DP} - 1)}{(PR_{DPrefmap} - 1)}(PR_{refmap} - 1) + 1 \tag{36}$$

$$Mass\ flow\ (W) = CMSF * W_{refmap} \tag{37}$$

$$Component\ Efficiency\ (\eta_c) = \eta_c SF * \eta_{c_{refmap}} \tag{38}$$

$$Work\ Function\ (DH) = DHSF * DH_{refmap} \tag{39}$$

At the moment, GT-ACYSS does not include maps for the heat exchanger; hence the pressure drops were scaled with mass flow relationship expressed below:

$$\Delta P_{OD} = \Delta P_{Dp} \times \left(\frac{W_{OD}}{W_{DP}}\right)^{1.75} \tag{40}$$

Similarly, the heat exchanger effectiveness is scaled as thus:

For inter-cooler and pre-cooler

$$\varepsilon_{OD} = 1 - (1 - \varepsilon_{DP}) \times \left(\frac{NDMF_{OD}}{NDMF_{DP}}\right) \tag{41}$$

For recuperator and gas heater

$$\varepsilon_{OD} = 1 - (1 - \varepsilon_{DP}) \times \left(\frac{W_{OD}}{W_{DP}}\right) \tag{42}$$

## 8. Control Options

At the moment, GT-ACYSS allows the user to select from a list of different control options according to reference plant configuration and performance assessment to be carried out. The control options include Inventory control mode, Bypass control mode, Heat source temperature control mode and combined mode (integration of the listed control modes to achieve different control task).

The operating logic for inventory control is that, for any changes in load demand or design point operating conditions, the working fluid is extracted or injected onto the power plant conversion system via the opening and/or closing of the inventory valves, resulting in changes in mass flow rate, and other properties of the cycle. During this operation, the heat source temperature and bypass remain constant. In the case of bypass control, mass flow bled off from the compressor discharge to short-circuit the turbine. This control is used for fast response to changes in power demand. The heat source temperature control alters the turbine entry temperatures based on changes in design point operating condition at constant mass flow inventory. The combined mode utilizes an integration of control options discussed above to regulate the behaviour of the engine in order to avoid limitations in shaft speed or low cycle efficiencies. A typical example is a combined control mode utilizing inventory control for slow load reduction and bypass control for rapid load response and shaft speed control. Further discussions on closed-cycle gas turbine control options are documented in reference [33–35].

## 9.Solution Convergence Method

The solution convergence technique requires the use of a control mechanism to satisfy mass flow, speed and power compatibility. The mathematical approach to this non-linear relationship between the dependent and independent variables required several iterations. The approach used in GT-ACYSS was essentially similar basic technique used in references [10,25–27,36] with minor modifications. The solution to the unknown independent variable which matches the computed value of the dependent variables to it's required or estimated values were solved using a combination of the secant-bisection method. The derivation of the method has been extensively discussed in reference [37]. Basically, the iterative process begins with a first pass through the entire cycle configuration made with initial guesses for all unknown independent variables determined by the control option selected (for example mass flow for inventory control and bypass, or Turbine Entry Temperature (TET) for heat source temperature control, compressor pressure ratio, heat exchanger efficiencies etc). The results of the first pass are checked through the compatibility matching and the succeeding passes through the configuration are made changing each respective unknown by a small amount from the convergence solution technique. The convergence tolerance in GT-ACYSS is $10^{-5}$. A typical illustration using engine shown in fig.3 to demonstrate the structure of simulation convergence in GT-ACYSS is described by fig. 2.

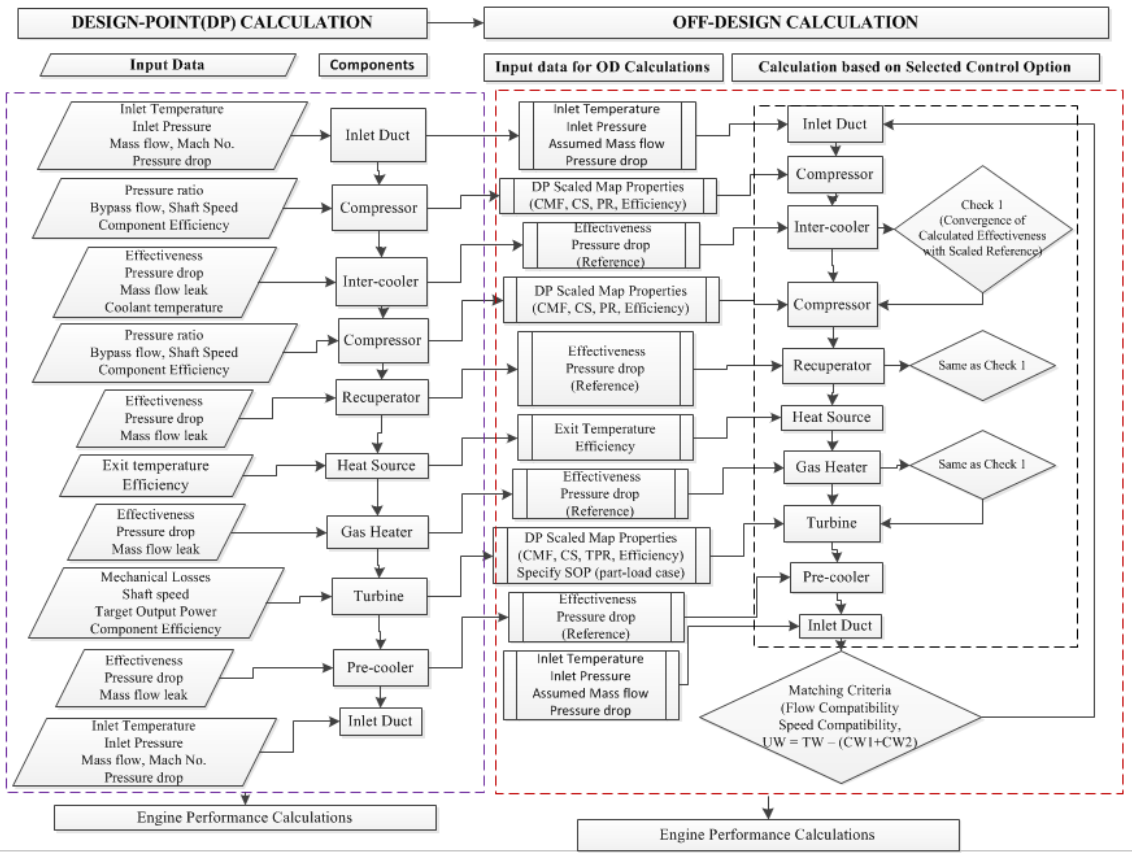


**Fig. 2 Structure of Simulation Convergence in GT-ACYSS**

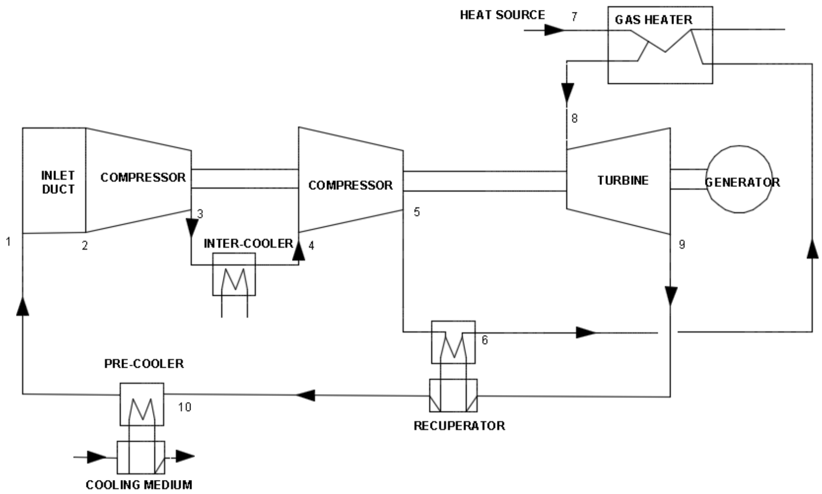


**Fig. 3 Schematic diagram of a closed-cycle plant used for the illustrative purpose in section (VI & VIII)**

## 10. Application of Simulation Tool

To verify the capability of the simulation tool, three closed-cycle gas turbine plants with reference data obtained from the open literature were adapted and performance simulated. The reference plants selected consist of two recuperated cycle plants and an intercooled-recuperated cycle plant shown in fig. 4, 5, and 6. The design point performance of all selected engine was modelled based on published design specifications and an optimal combination of both known and guessed parameters was utilized to obtain reasonable minimum mean deviations of performance and major gas path parameters from the published reference data. After the design point specifications were simulated, off-design performance variation was predicted and verified with available reference data using appropriate control handle mentioned in section VII.

## 11. Results and Discussion

### A. Case 1

The modelled plant for this case is a single-shaft recuperated closed-cycle configuration shown schematically in Figure 4, inspired on GTHTR300. The plant under development by the Japan Atomic Energy Agency (JAEA) is powered with a nuclear reactor cooled with helium. It consists of a single turbomachinery set (turbine-compressor), a recuperator, pre-cooler and the reactor. The performance of the engine was simulated with GT-ACYSS and results at design point compared with data in reference [38–41].

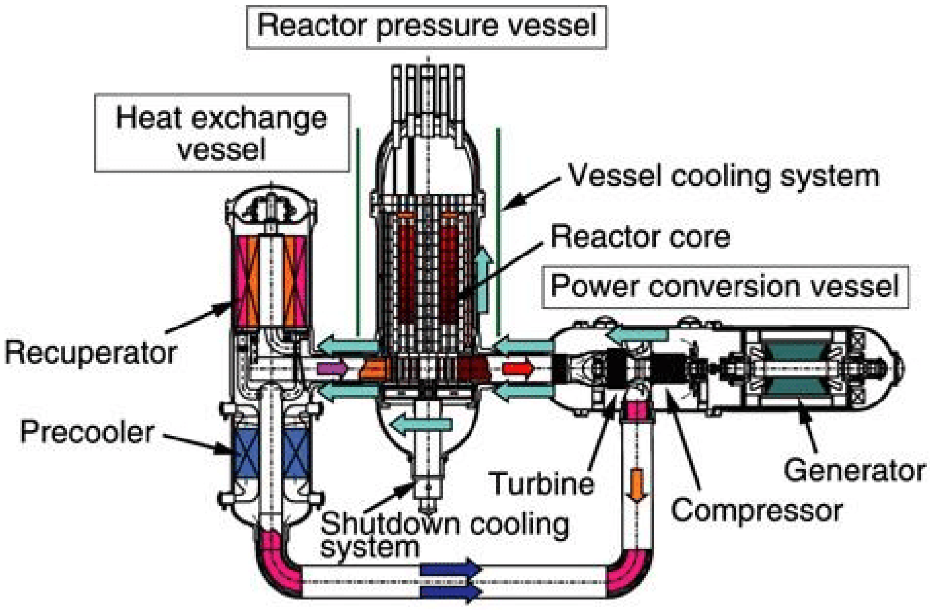


**Fig. 4 Schematic of the reference plant modelled for case 1** [38–41]

During the simulation, total pressure losses accounted for comprises of (1.7% at the recuperator, and 1.4% at Reactor), Bypass cooling of 0.5% and mass flow leakage of 1% were estimated based on knowledge of the main design parameters. Table 2 shows a detailed comparison of the simulated performance of the engine with published data. The predictions of the component performance characteristics and gas path parameters fell within close range compared with available information. Hence, this enabled a feasible simulation of the engine overall performance indicators such as power generated, cycle efficiency and Exhaust Gas Temperature (EGT); which were in reasonable agreement with the reference data. A summary of the deviation of predicted performance to reference data is shown in Table 2. In general, the average deviation of the simulated values from reference design point data was 0.39%. This means that the aim of producing the engine performance parameters through accurate prediction of component characteristics parameters can be realized reasonably using GT-ACYSS.

**Table 2. Comparison of Simulated results and Published Design Point Data for case 1**

| **Description** | **Reference Plant DP (Published Data)** [38–41] | **GT-ACYSS (Simulated)** | **Deviations (%)** |
|---|---|---|---|
| Turbine entry temp. (°C) | 850 | 850 | - |
| Shaft Speed (rpm) | 3600 | 3600 | - |
| Compressor pressure ratio | 2.0 | 2.0 | - |
| Compressor inlet Pressure (MPa) | 3.5 | 3.5 | - |
| Compressor inlet temp. (°C) | 28 | 28 | - |
| Compressor polytropic efficiency (%) | 90.5 | 89.9 | -0.6 |
| Reactor power (MW) | 600 | 603 | +0.5 |
| Turbine polytropic efficiency (%) | 93 | 91.8 | -1.3 |
| Turbine pressure ratio | 1.88 | 1.90 | +1.0 |
| Turbine EGT (°C) | 611 | 613 | +0.2 |
| Flow rate at compressor (kg/s) | 441.8 | 441 | -0.2 |
| Plant thermal efficiency (%) | 46.8 | 46.9 | +0.2 |
| PC effectiveness (%) | 95 | 95.0 | - |
| ReX effectiveness (%) | 95 | 94 | -1.0 |
| Rated power (MW) | 280 | 283.1 | +1.1 |
| Working Fluid | Helium | | |

Next, the accuracy of predicting the engine compressor map and off-design condition at part-load was simulated and compared with results in reference [41]. To achieve this, the reference map was digitized and replotted on the non-dimensional basis for easy comparison as shown in fig. 5. Figure 5 shows simulated results of the pressure ratio and corrected mass flow as a function of rotational speed. As the engine throttles from 0% to design capacity speed, both the pressure ratio and corrected mass flow moves to the right with an increase in their values. Similarly, the heat input at full load to part-load condition was predicted by varying the compressor inlet pressure at constant turbine entry temperature. Figure 6 describes the performance at part load using inventory control.

Comparing the simulated results to reference data showed quite a reasonable agreement, which confirms the validity of the program off-design simulation routine. For the compressor map, six-speed lines of the simulated result were analyzed and their root-mean-square deviations calculated. The largest root mean square obtained was less than 0.5% which confirms that the results were quite in good agreement with reference data.

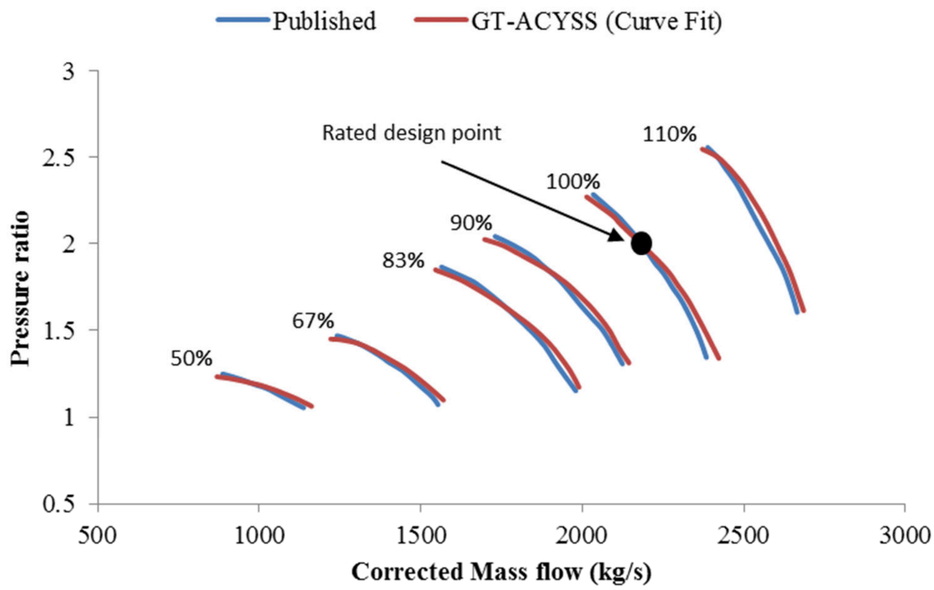


**Fig. 5 Compressor performance comparisons with GT-ACYSS at full speed**

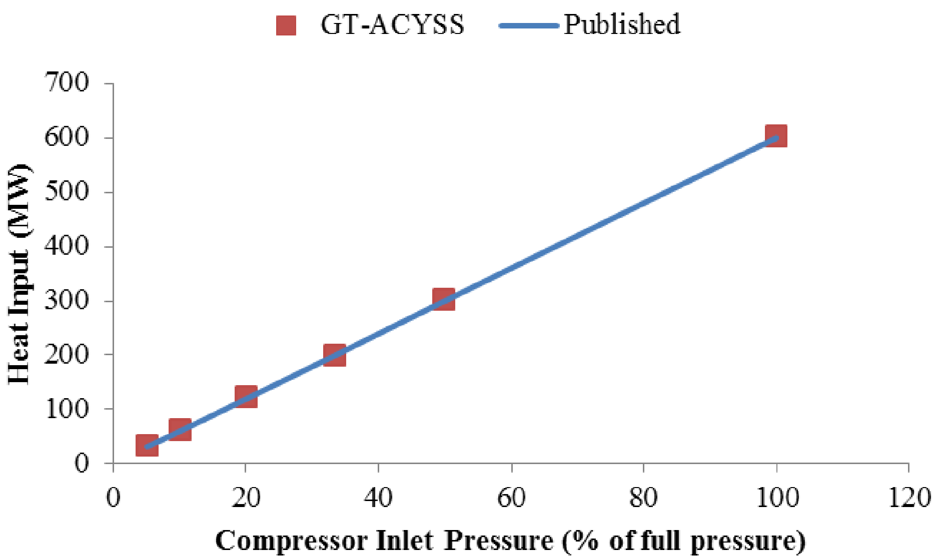


**Fig. 6 Heat power requirement comparisons from full to part-load**

## B. Case 2

The reference power plant simulated in case 2 is an intercooled-recuperated closed-cycle gas turbine inspired by the Escher Wyss Ravensburg power plant in Germany. Its main design characteristics were based on the cycle shown in reference [42]. The cycle consists of a low-pressure compressor (LPC), intercooler (IC), high-pressure compressor (HPC), recuperator heat exchanger (RX), gas-heater (GH), turbine, and a pre-cooler (PC). Both the LPC and HPC are driven on a single shaft. The engine is run on coal. Total system pressure loss of 8%, mechanical losses to drive compressors at 10% and total flow leakage of 0.5% were considered during simulation. The design and operational data used for comparison were obtained from reference [42].

The result of simulation at the design point was compared with published data and the deviation calculated. A summary of the engine performance is presented in table 3. Again, the results of simulated performance parameters were in good agreement with the reference data with the largest deviation at 1.7%. In general, an average deviation of 0.45% was obtained from the predicted performance.

Fig. 7 shows an off-design performance prediction at changes in compressor inlet temperature due to changes in the cooling temperature. To achieve the simulation goal, the compressor inlet temperature is used as an input. As compressor inlet temperature increases, it changes the position of $N/\sqrt{T_1}$ and at constant TET, the mass flow inventory is increased to compensate for the drop in power, since the engine was design to operate at constant power output, mass flow increase cause the system pressure to increase as shown in fig. 7. This change causes both the LPC and HPC pressure ratio to drop thereby reducing the cycle thermal efficiency. Although, there was no reference data to compare with the simulated result, however, the performance trend at varying condition seems to be in agreement with the behavior of closed-cycle gas turbine at changes in compressor inlet temperature.

**Table 3. Comparison of Simulated results and Published Design Point Data for case 2**

| **Description** | **Reference Plant DP (Published Data)** [42,43] | **GT-ACYSS (Simulated)** | **Deviations (%)** |
|---|---|---|---|
| Turbine entry temp. (℃) | 662 | 663.0 | +0.11 |
| Shaft speed (rpm) | 3000 | 3000 | - |
| LPC pressure ratio | 1.65 | 1.64 | -0.60 |
| LPC inlet pressure (MPa) | 0.8 | 0.80 | - |
| LPC inlet temp. (℃) | 17 | 17 | - |
| HPC Pressure ratio | 2.40 | 2.40 | - |
| LPC & HPC Isentropic efficiency (%) | 78.50 | 78.00 | -0.60 |
| Heat input (MW) | 9.87 | 9.88 | +0.10 |
| Turbine Isentropic efficiency (%) | 88 | 88.30 | +0.30 |
| Turbine pressure ratio | 3.64 | 3.65 | +0.20 |
| Turbine EGT (℃) | 419.15 | 427.00 | +1.10 |
| Flow rate at Compressor (kg/s) | 27.83 | 27.83 | - |
| Plant thermal efficiency (%) | 23.30 | 23.68 | +1.63 |
| IC effectiveness (%) | 90.00 | 90.50 | +0.50 |
| PC, RX & GH effectiveness (%) | 84.70 | 85.00 | +0.35 |
| Rated power (MW) | 2.3 | 2.34 | +1.74 |
| Working Fluid | Air | | |

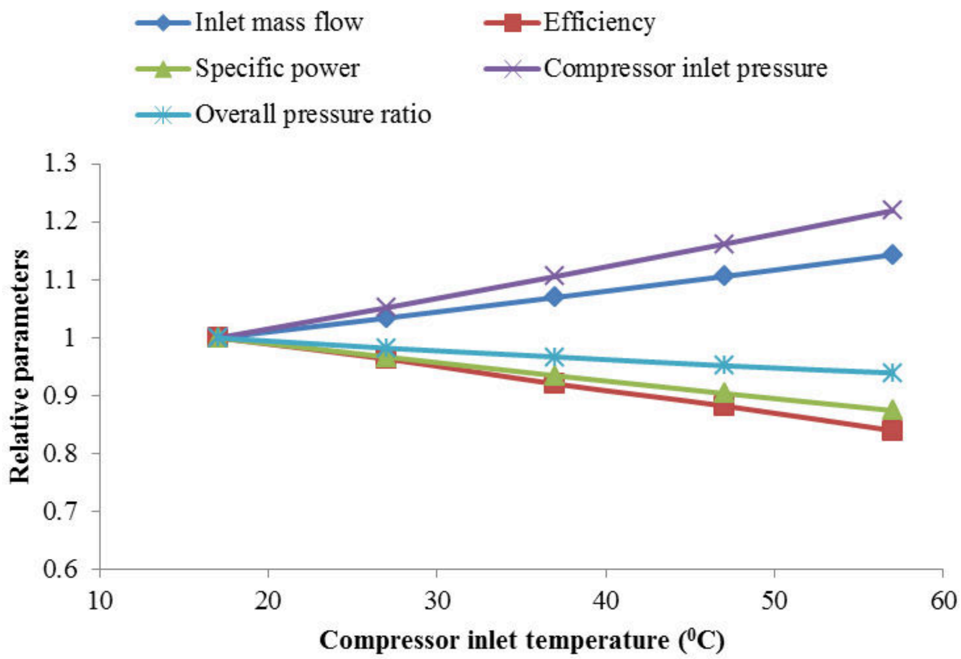


**Fig. 7 Variation in performance due to changes in compressor inlet temperature for fixed power operation**

### C. Case 3

The major components of case 3 plants consist of a turbine-compressor set, a recuperator, and a pre-cooler as shown in fig. 8. The thermal energy is provided by the naturally-fired gas heater and nitrogen was used as working fluid. The engine control system is based on maintaining a constant working fluid inventory in the system at constant turbine inlet temperature and controlling power output and speed by bypassing working fluid from the compressor discharge into the low-pressure heat exchanger. The design point and off-design simulation of the reference engine were carried out in GT-ACYSS and compared with data from open literature [6,24,44]. All units during simulation design and off-design were converted to the International System of Units (SI), hence, the reference maps and performance parameters were replotted.

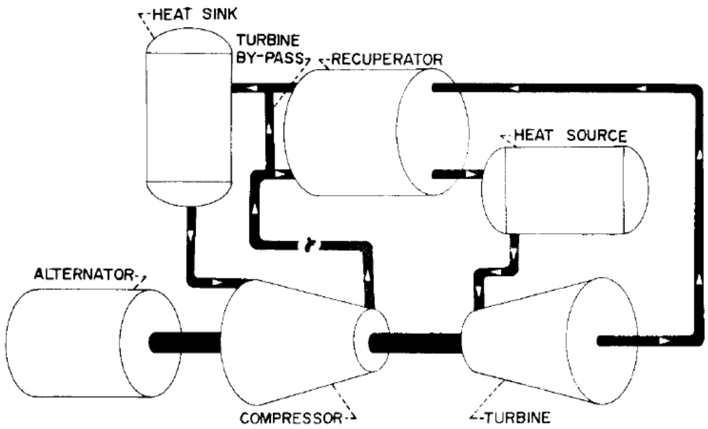


**Fig. 8 Schematic of reference plant modelled for case 3** [6]

Table 4 shows the performance comparison of simulated results with reference data at the design point. The deviation may be due to the influence of the gas properties but are within reasonable limits with the largest deviation at 2.5% and an average deviation of 0.86%. Figure 9 and 10 are the off-design performance of the engine compared with reference data. The off-design part-load presented in fig. 9 was simulated using bypass control. This means that the working fluid is bled off to short-circuit the gas heater (GH) and turbine into the low-pressure side of the recuperator, and, as such, the mass flow, pressure ratio and efficiency of the turbine drop, causing a decrease in the output power. This control option is usually implemented in closed-cycle gas turbines to achieve fast control response on power drop and to prevent the shaft from over-speed.

**Table 4. Comparison of Simulated results and Published Design Point Data for case 3**

| **Description** | **Reference Plant DP (Published Data)** [6,24,44] | **GT-ACYSS (Simulated)** | **Deviations (%)** |
|---|---|---|---|
| Turbine entry temp. (℃) | 649 | 648.5 | +0.05 |
| Shaft speed (rpm) | 22000 | 22000 | - |
| Compressor pressure ratio | 2.72 | 2.72 | - |
| Compressor inlet pressure (MPa) | 0.8 | 0.8 | - |
| Compressor inlet temperature (℃) | 55 | 55 | - |
| Compressor Isentropic efficiency (%) | 82 | 80.40 | -1.95 |
| Heat input (MW) | 3.058 | 3.133 | +2.45 |
| Turbine isentropic efficiency (%) | 86 | 84.70 | -1.50 |
| Turbine pressure ratio | 2.38 | 2.36 | -0.84 |
| Flow rate at Compressor (kg/s) | 11.8 | 11.10 | -2.50 |
| Plant thermal efficiency (%) | 16.7 | 16.3 | -2.40 |
| PC effectiveness (%) | 94 | 94.0 | - |
| ReX effectiveness (%) | 79.0 | 78.8 | -0.25 |
| Rated power (MW) | 0.510 | 0.511 | +0.23 |
| Working Fluid | Nitrogen | | |

The result in fig. 9 shows a close prediction of part-load performance compared to published data, with a root mean square deviation of 0.16%. Figure 10 shows an increase of pressure ratio and corrected mass flow as the engine speed is throttled from idle to synchronous idle at zero power [45], which is consonance to the behaviour of closed-cycle gas turbines. This result demonstrates the capability of the code to simulate or monitor engine characteristics during idling.

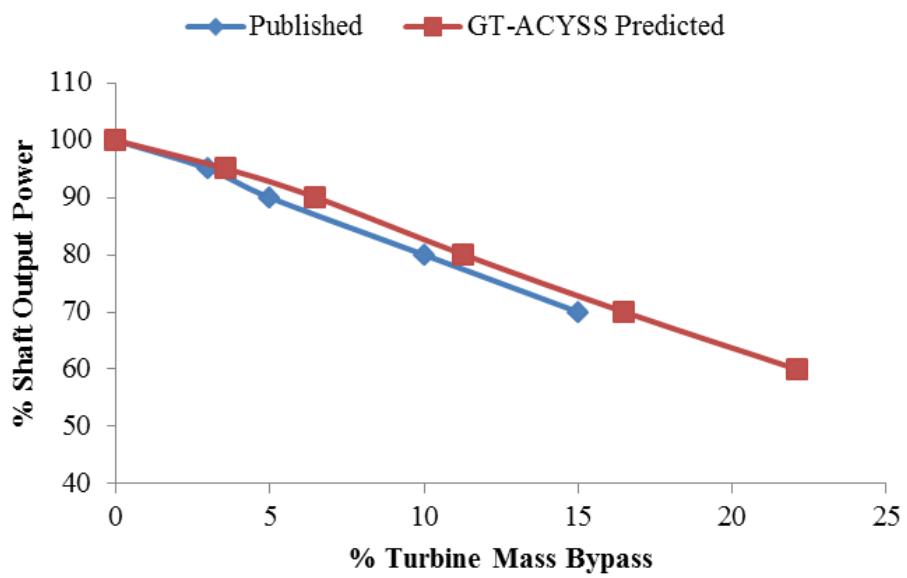


**Fig. 9 Part Load Performance Comparison**

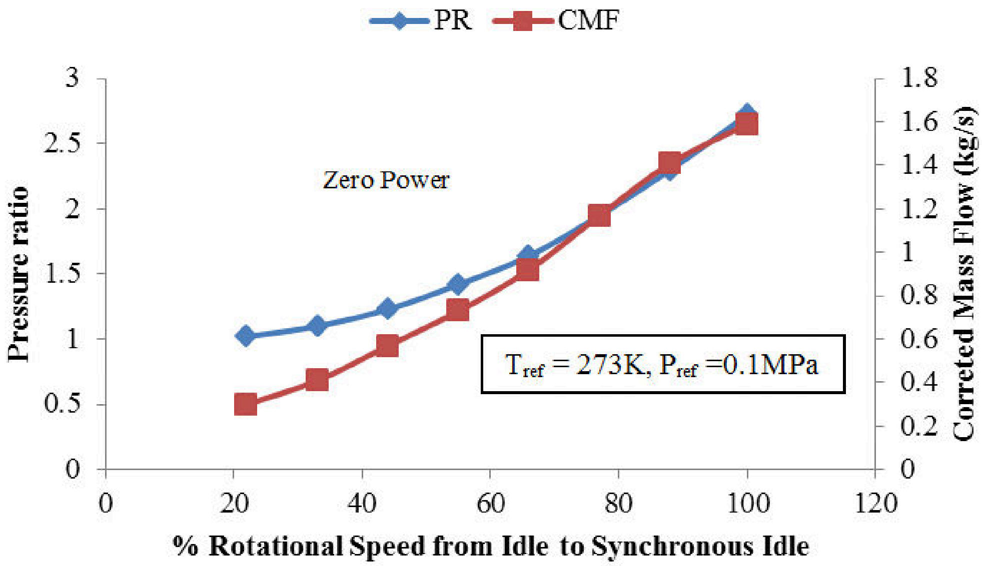


**Fig. 10 Engine Rotational Speed as a function of pressure ratio and corrected mass flow (kg/s)**

## 12. Conclusion

This paper described the procedure that has been used to develop GT-ACYSS; a performance simulation for closed-cycle gas turbines. The main concluding features and advantages of the simulation tool presented are:

- Components can be assembled to represent different single-shaft engine configurations; therefore it should prove to be useful for exploratory studies at design stage based on specified operational requirement; which can save the cost of manufacturing or test an expensive prototype.
- Four different fluids have been incorporated to give the user flexibility and users can either do performance calculation with or without component maps. Incorporating different working fluid models allows the user to compare influence working on the cycle performance and component designs.
- The simulation tool can be used to investigate the effect of components performance characteristics on the overall performance of the closed-cycle engine and the capability to carry out preliminary component design.
- The physical behaviour of the system components has been described using analytical and empirical equations. The solution of these equations under varying operating conditions defines the performance and behaviour of the reference engine in response to prescribed input parameters.
- The results presented herein demonstrate that the scaling method of component maps and the algorithm of steady-state analysis were reasonable with mean deviations for different case studies between 0.1 to 1%.
- Although it has not been possible to compare results with test data, the trends shown in case studies appear to be acceptable when compared with results in the open literature.
- The engine model, although it shows a good correlation with the expected performance trend, there are several limitations on the modelling technique that could be improved. Firstly, the component maps utilized are scaled from maps within the GT-ACYSS libraries for the specified working fluid, which is very unlikely to be the same as the real engine map. Secondly, the overall pressure ratio was modelled based on data available in the open literature, however, for cases that required several compressors, the distribution of the pressure ratio amongst each compressor was made based on assumptions and technical judgment. Finally, the TETs were chosen so that the required output characteristics are met, however, the real TET may be different, which could slightly affect the overall cycle performance.
- Modelling of transient behaviour of the closed-cycle gas turbine is another area of improvement that would be incorporated into the cycle performance simulation tool in future. Also, further modification to

accommodate complex cycle configuration and component maps would be implemented into the GT-ACYSS model.

## Appendix

| | | | |
|---|---|---|---|
| **Correlation Coefficient for Equation 1 at 1atm** | | | |
| $A_0$ = 4.2360E+02 | $A_1$ = 1.8618E00 | $A_2$ = -1.8068E-03 | $A_3$ = 1.0375E-06 |
| $A_4$ = -3.2606E-10 | $A_5$ = 4.3032E-14 | $A_6$ = - 1.5604E-19 | |
| **Correlation Coefficient for Equation 2 at 10atm** | | | |
| $A_1$ = 6.1387E+02 | $A_2$ = 5.8227E-01 | $A_3$ = - 2.5931E-04 | $A_4$ = 6.9978E-08 |
| $A_5$ = -9.6851E-12 | $A_6$ = 3.8926E-16 | | |

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2019-05-08

# GT-ACYSS: gas turbine arekret-cycle simulation modelling for training and educational purposes

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Osigwe E, Pilidis P, Nikolaidis T, Sampath S. (2019) GT-ACYSS: gas turbine arekret-cycle simulation modelling for training and educational purposes. Journal of Nuclear Engineering and Radiation Science, Volume 5, Issue 4, October 2019, Article number 041207

https://doi.org/10.1115/1.4043681